Journal of Engineering Physics and Thermophysics, Vol. 81, No. 2, 2008

CHARACTERISTIC FEATURES OF THE PROCESS OF MIXTURE FORMATION UPON FUEL INJECTION INTO A HIGH-TEMPERATURE AIR FLOW

A. I. Maiorova and V. V. Tret'yakov

UDC 532.529.5:536.24

Physical models of the processes of interphase heat and mass transfer and computational techniques based on them are suggested. The problem of the interaction of fuel droplets with a high-temperature air flow is solved numerically. It is shown that injection of a fuel by a spray atomizer may cause a substantial change in the gas temperature. With pneumatic spraying of a fuel by a cold air jet the influence of interphase exchange on the characteristics of the main flow is insignificant. Upon injection of a hot spraying air the influence of fuel injection on the formation of temperature fields is more appreciable. However, here the gas is cooled less than in the case of spraying by a jet.

Introduction. Combustion of a fuel in the combustion chambers of a gas-turbine engine and a gas-turbine plant is closely connected with the processes of mixing [1]. Investigations of these processes carried out by both experimental and computational methods [1–3] have recently become especially crucial because of the necessity of solving ecological problems.

One of the most pressing problems at present is account for the influence of droplets on an air flow. In some of the regimes of chamber operation this may lead to a substantial, almost twofold, change in the long range of a fuel spray and, consequently, to corresponding changes in the distributions of the concentrations of fuel phases.

In this work we set out to calculate the fields of air velocity and temperature as well as of the distribution of a liquid fuel in a module combustion chamber with account for the processes of heating and evaporation of droplets in those regimes typical of combustion chambers in which there is a substantial interphase exchange. The influence of interphase heat exchange on mixing in the working volume of the module was considered earlier in [4]. It was shown that the computational technique describes well the experimental data on the configuration of the fuel spray. In the present work, the technique is supplemented with account for mass exchange between liquid and gas phases.

It is clear that when a "cold" fuel is supplied into a "hot" air flow, the droplets are heated and the air surrounding them is cooled. It is evident that at small flow rates of the fuel this cooling can be neglected. The aim of this work is to answer two questions: how much the air flow is cooled in the range of parameters typical of real combustion chambers, and how far the region of flow cooling extends. Moreover, the dependence of the flow characteristics on the means of fuel spraying (jetty or pneumatic) and also on the spraying air temperature is investigated.

Statement of the Problem. The model selected for investigation is a straight channel of rectangular cross section 50 × 30 mm and 150 mm long into which air is supplied at a velocity U_0 and temperature T_0. Injection of a fuel with a temperature T_f into the channel at a velocity V_f is made through a hole in the upper wall of the channel with the aid of an injector installed along the normal to the longitudinal axis of the channel halfway between the side walls. In modeling the pneumatic injector it is considered that, coaxially with the fuel supply, the spraying air is fed at a velocity U_1 and temperature T_1 into the channel through a rectangular hole of size 4.5 × 3.75 mm. In modeling a jetty injector, we assume that the spraying air is absent. Calculations were carried out for the velocity and temperature of the main air flow $U_0 = 20$ m/sec and $T_0 = 900$ K, $V_\mathrm{f} = 8$ m/sec, $T_\mathrm{f} = 300$ K. The gas pressure at the channel inlet was equal to 100 kPa. The variable parameters of the calculation were the velocity and temperature of the spraying air: $U_1 = 0$–20 m/sec and $T_1 = 300$–900 K, as well as the summed coefficient of air excess through the module $\alpha = 1.35$–5.4. The values of the regime parameters are presented in Table 1. Regime 1 corresponds to jet spraying of

---

P. I. Baranov Central Scientific Research Institute of Aircraft Engines, 2 Aviamotornaya Str., Moscow, 111116, Russia; email: majorova@ciam.ru. Translated from Inzhenerno-Fizicheskii Zhurnal, Vol. 81, No. 2, pp. 250–258, March–April, 2008. Original article submitted May 23, 2006; revision submitted September 12, 2007.

TABLE 1. Operating Parameters

Variant	α	Regime 1	Regime 2		Regime 3	
		U_1, m/sec	U_1, m/sec	T_1, K	U_1, m/sec	T_1, K
1	5.4	0	20	300	20	900
2	2.7	0	20	300	20	900
3	1.35	0	20	300	20	900

a fuel, regime 2 — to pneumatic spraying of a fuel by a cold air jet; and regime 3 — to pneumatic spraying by a hot air jet in the limiting case of equality between the temperatures of the spraying air and main flow.

Calculation Technique. Calculations of the flow of a gas phase are based on numerical integration of the full system of stationary Reynolds equations and total enthalpy conservation equations written in Euler variables. The technique of allowing for the influence of droplets on a gas flow is based on the assumption that such an allowance can be made by introducing additional summands into the source terms of the mass, momentum, and energy conservation equations. A more detailed description of the procedure can be found in [5]. The transfer equations were written in the following conservative form:

$$\operatorname{div}\left(\rho_g \mathbf{U}_g \varphi - \Gamma_\varphi \operatorname{grad} \varphi\right) = S_\varphi + S_\varphi^{\text{int}} . \tag{1}$$

Here S_φ^{int} is the interphase source term that describes the influence of droplets on the corresponding characteristics of flow. The density and pressure are ensemble-averaged (according to Reynolds) and all the remaining dependent variables — according to Favre, i.e., with the use of density as a weight coefficient.

Written in the form of Eq. (1), the system of equations of continuity ($\varphi \equiv 1$, $\Gamma_\varphi \equiv 0$, $S_\varphi \equiv 0$), motion ($\varphi = U_{gi}$, $i = 1, 2, 3$), and of total enthalpy conservation h ($S_h \equiv 0$) is solved by the simple finite-difference iteration method [6]. The walls were considered thermally insulated. To find the coefficients of turbulent diffusion, use is made of the Boussinesq hypothesis on the linear dependence of the components of the tensor of turbulent stresses on the components of the tensor of deformation rates of average motion and two equations of transfer of turbulence characteristics (k–ε) in the modification that takes into account the influence of flow turbulence Reynolds numbers on the turbulent characteristics of flow [7]. Here, the boundary conditions of zero velocity are imposed on the solid walls.

In the absence of chemical reactions the gas mixture is considered to consist of two components: kerosene vapors (with a molecular weight of 0.168 kg/mole) and air (with a conventional molecular weight of 0.029 kg/mole). For the mass fraction of kerosene vapors m_f the equation of transfer of the type (1) is solved, and the mass fraction of air is determined from the condition under which the sums of the mass fractions of all the components are equal to unity.

The calculations of the distribution of fuel are based on the solution of a system of equations of motion, heating, and evaporation of individual droplets written in the Lagrange variables [8]. The influence of turbulent pulsations onto the motion of droplets and on the change in their shape in the process of their motion is considered to be negligibly small. Then the equations that describe the processes of motion, heating, and evaporation have the following form:

$$m_d \frac{d\mathbf{V}_d}{dt} = \mathbf{R} , \tag{2}$$

$$\frac{dT_d}{dt} = 6\lambda_g \left(1 + 0.6\mathrm{Re}^{1/2}\mathrm{Pr}^{1/3}\right) \frac{T_g - T_d}{\rho_d D_d^2 c_f} , \tag{3}$$

$$\frac{dD_d}{dt} = -\frac{8\lambda_g}{2D\rho_d c_{pg}} \ln\left(1 + \frac{c_{pg}}{L}(T_g - T_d)\right)\left(1 + 0.23\mathrm{Re}^{1/2}\right). \tag{4}$$

We consider that the law of the resistance of droplets is the same as that of the resistance of solid spherical particles of diameter D_d:

$$\mathbf{R} = 0.5 C_R S \rho_\text{g} W \mathbf{W}\,, \quad C_R = 24\text{Re}^{-1} + 4.4\text{Re}^{-0.5} + 0.32\,, \quad S = \pi D_\text{d}^2/4\,.$$

In modeling a fuel spray it was assumed that it had a polydisperse structure with the size distribution of droplets obeying the Rosin–Rammler law [9] with exponent 3 and mean-median diameter 50 μm. The range of the sizes of droplets was divided into 14 intervals. The angle distribution of droplets was taken to be uniform, and the working fuel was TS-1 kerosene.

The interphase source terms are calculated together with the distribution of the liquid fuel from the conditions of the fulfillment of the laws of conservation of momentum, mass, and heat of the gas–droplet system. It is considered that the corresponding terms in the equations for the turbulence characteristics can be neglected.

Since physically the source term S_1^int in the continuity equation, just as the source term in the equation of transfer of m_f, $S_{m_\text{f}}^\text{int}$, is the increase in the concentration of the fuel vapor per unit time equal to the rate of liquid evaporation, then

$$S_1^\text{int} = S_{m_\text{f}}^\text{int} = \dot{C}_\text{v} = -\frac{dC_\text{f}}{dt}\,, \tag{5}$$

where $\dot{C}_\text{v}$ is the rate of change of C_v due to the interphase exchange.

The interphase source terms in the equations of conservation of momentum components are the components of the vector of the rate of change in the gas momentum due to the exchange with droplets in a unit volume $\mathbf{S}_\text{g}^\text{int}$. These quantities are determined from the equation of conservation of momentum for the gas–droplet system:

$$\Delta\,(m_\text{d}\mathbf{V}_\text{d}) + \Delta\,(m_\text{g}\mathbf{U}_\text{g}) = 0\,, \tag{6}$$

where m_g is the mass of the isolated element of the gas volume Δv. Here and below, it is assumed that the volume of fuel droplets is negligibly small as compared to the volume occupied by the gas.

Assuming Δt_d (the residence time of a droplet in the volume element Δv) to be small enough, we may replace the second term in (6) by $\mathbf{S}_\text{g}^\text{int}\Delta v \Delta t_\text{d}$. This gives us an approximate expression to determine $\mathbf{S}_\text{g}^\text{int}$:

$$\mathbf{S}_{\text{g}i}^\text{int} = \frac{\Delta\,(m_\text{g}\mathbf{U}_\text{g})}{\Delta v \Delta t_{\text{d}i}} = -\frac{m_{\text{d}i}}{\Delta v}\frac{\Delta \mathbf{V}_{\text{d}i}}{\Delta t_{\text{d}i}} - \mathbf{V}_{\text{d}i}\frac{\Delta C_{\text{f}i}}{\Delta t_{\text{d}i}}\,, \tag{7}$$

where ΔV_d is a change in the droplet velocity during its residence in the elementary volume.

The last term in relation (7) describes the gas momentum increment at the expense of the vapor fuel phase momentum related to the elementary volume Δv and the time of droplet evaporation in this volume, since $\Delta C_\text{v} = -\Delta C_\text{f}$. It is assumed that the fuel vapor and air in the volume Δv mix instantaneously. When $\Delta v \to 0$, $\Delta t_\text{d} \to 0$, we obtain an exact expression for $\mathbf{S}_\text{g}^\text{int}$ in a differential form:

$$\mathbf{S}_{\text{g}i}^\text{int} = -C_{\text{f}i}\frac{d\mathbf{V}_{\text{d}i}}{dt} - \mathbf{V}_{\text{d}i}\frac{dC_{\text{f}i}}{dt}\,. \tag{8}$$

Here $C_{\text{f}i}$ denotes a fraction of the ith droplet in the volumetric concentration of liquid.

The summed value of the rate of change in the momentum of a unit volume of gas is equal to

$$\mathbf{S}_{\mathrm{g}}^{\mathrm{int}} = \sum_i \mathbf{S}_{\mathrm{g}i}^{\mathrm{int}}, \tag{9}$$

where summation is carried out over all the droplets. The quantity $\Delta\mathbf{V}_{\mathrm{d}i}/\Delta t_{\mathrm{d}i}$, or $d\mathbf{V}_{\mathrm{d}i}/dt$ is taken from the solution of the equation of motion of an individual droplet.

The interphase source term in the equation for enthalpy S_h^{int} that describes heat exchange between droplets and the gas flow is determined from the equation of conservation of the total enthalpy of the gas–droplet system, which has the form

$$\Delta\,(m_{\mathrm{d}}h_{\mathrm{d}}) + \Delta\,(m_{\mathrm{g}}h_{\mathrm{g}}) = -\,L\Delta m_{\mathrm{g}}\,. \tag{10}$$

The expression on the right-hand side of equality (10) determines the energy spent on the transition of the droplet liquid of mass $\Delta m_{\mathrm{d}} = -\Delta m_{\mathrm{g}}$ into the gaseous state, and Δh_{d} and Δm_{d} are changes in the enthalpy and mass of the droplet during its residence in the volume Δv.

Assuming the time Δt_{d} to be small enough, we replace the second term in expression (8) by $S_h^{\mathrm{int}}\Delta v\Delta t_{\mathrm{d}}$. Then the approximate expression for determining $S_{h,i}^{\mathrm{int}}$ has the form

$$S_{h,i}^{\mathrm{int}} = \frac{\Delta\,(m_{\mathrm{g}}h_{\mathrm{g}})}{\Delta v\Delta t_{\mathrm{d}i}} = -\frac{\Delta\,(m_{\mathrm{d}i}h_{\mathrm{d}i})}{\Delta v\Delta t_{\mathrm{d}i}} + L\frac{\Delta m_{\mathrm{d}i}}{\Delta v\Delta t_{\mathrm{d}i}} \tag{11}$$

or (provided the mass of the droplet changes only at the expense of evaporation)

$$S_{h,i}^{\mathrm{int}} = L\frac{\Delta C_{\mathrm{f}i}}{\Delta t_{\mathrm{d}i}} - C_{\mathrm{f}i}\frac{\Delta h_{\mathrm{d}i}}{\Delta t_{\mathrm{d}i}}\,. \tag{12}$$

Using the definition of the enthalpy $h_{\mathrm{d}} = c_{\mathrm{f}}T$, we will rewrite (12) in the form

$$S_{h,i}^{\mathrm{int}} = L\frac{\Delta C_{\mathrm{f}i}}{\Delta t_{\mathrm{d}i}} - C_{\mathrm{f}i}\left(c_{\mathrm{f}}\frac{\Delta T_{\mathrm{d}i}}{\Delta t_{\mathrm{d}i}} + T_{\mathrm{d}i}\frac{\Delta c_{\mathrm{f}i}}{\Delta t_{\mathrm{d}i}}\right). \tag{13}$$

When $\Delta v \to 0$, $\Delta t_{\mathrm{d}} \to 0$, we obtain an expression for $S_{h,i}^{\mathrm{int}}$ in a differential form:

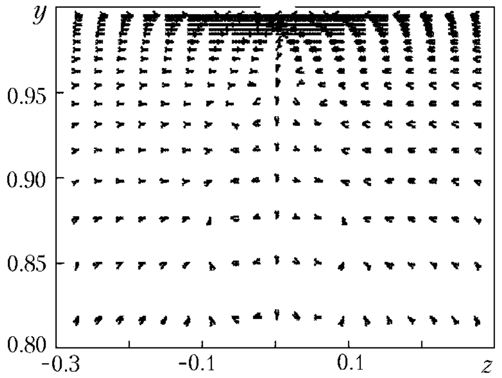
$$S_{h,i}^{\mathrm{int}} = L\frac{dC_{\mathrm{f}i}}{dt} - C_{\mathrm{f}i}\left(c_{\mathrm{f}}\frac{dT_{\mathrm{d}i}}{dt} + T_{\mathrm{d}i}\frac{dc_{\mathrm{f}}}{dt}\right). \tag{14}$$

The summed value of S_h^{int} (inflow of heat from the liquid phase to the unit volume of gas) is equal to

$$S_h^{\mathrm{int}} = \sum_i S_{h,i}^{\mathrm{int}}, \tag{15}$$

where summation is carried out over all the droplets, and the values $\left(\frac{\Delta T_{\mathrm{d}}}{\Delta t}\right)_i$ and $\left(\frac{\Delta C_{\mathrm{f}}}{\Delta t}\right)_i$ or $\frac{dT_{\mathrm{d}i}}{dt}$ and $\frac{dC_{\mathrm{f}i}}{dt}$ are taken from the solution of the equation that describes the heating of an individual droplet.

The technique of calculation of a two-phase flow is based on the solution of a conjugate problem of flow of the gas and liquid media and heat exchange between them. First the problem of the motion of a gas is solved without account for the influence of the motion of droplets on the flow and then, based on the velocity and temperature fields obtained, the distribution of the liquid fuel is calculated as well as the interphase source terms. At the second stage, the gasdynamic and temperature fields are recalculated with account for the interphase sources (the results of the first



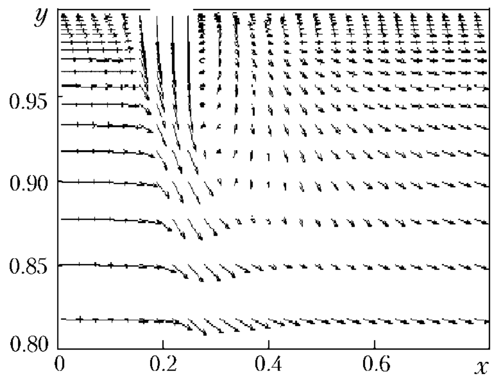


Fig. 1. Calculated vector velocity field in the cross section of the mixer $x = 0.28$ with jetty supply of fuel (regime 1, $U_1 = 0$); $\alpha = 1.35$.

Fig. 2. Calculated vector velocity field in the central longitudinal section of the mixer with pneumatic supply of a fuel — spraying by a hot air jet (regime 3, $U_1 = 20$ m/sec, $T_1 = 900$ K).

stage are used as the initial conditions). When needed, the process is repeated several times. The convergence criteria of the iteration process are considered to be the absence of changes in the velocity and temperature fields from iteration to iteration for the gas flow and stabilization over the iterations of the coordinate of the maximum value of the concentration of droplets at the outlet of the model within the limits of one mesh of the finite-difference grid.

As a result of methodical calculations a finite-difference grid uniform in the x and z directions was selected. The grid along the y axis was made finer toward the channel walls according to the exponential law with exponent 0.91. The total number of nodes in the grid was $111 \cdot 111 \cdot 41 = 505{,}161$. It was assumed that the stalling air flow at the inlet had a developed turbulent profile and that the spraying air had a uniform profile.

Results of Calculations. The results of calculation of the velocity field in the vicinity of the place of fuel injection for the jetty ($U_1 = 0$) and pneumatic ($U_1 = 20$ m/sec) sprayings are given in Figs. 1 and 2, respectively. Here and below, the results were made nondimensional through division by the characteristic dimension $H = 50$ mm, which is the height of the channel, and by the characteristic velocity $U_0 = 20$ m/sec.

In the absence of fuel supply at $U_1 = 0$ the flow is homogeneous and isothermal. In the case of the jetty spraying, as a result of the interaction of droplets with the main air flow, on both sides of the center of the injection hole, zones of reverse flow initiated by droplets are observed (Fig. 1), which increase with the fuel flow rate. At the same time, the very values of the secondary flow velocities are almost an order of magnitude smaller than the characteristic flow velocity. In the longitudinal section the shape of the velocity profiles preserves its inlet configuration.

In supplying spraying air (Fig. 2) the main role in the formation of the gas velocity fields is played by the interaction of air streams of the main and spraying air. Thus, behind the injected jet a secondary flow is formed in the form of a three-dimensional zone of reverse flows. The influence of the process of interaction of droplets with air on the flow structure is practically unnoticeable for the cases considered (the patterns of flow for all the regimes are practically identical). Moreover, the depth of penetration of fuel-air jets into the stalling air flow decreases with increase in the temperature of the spraying air due to the decrease in the injected gas momentum.

Figures 3–5 present the distributions of dimensionless volumetric concentrations of a liquid fuel c_f. The results were made nondimensional through division by the value of the main air flow density at the inlet.

A comparative analysis of the concentration fields in jetty spraying for various values of α has shown that for the higher values of the fuel flow rate there corresponds a wider fuel spray. This spray is more extended, its inner region is characterized by higher values of concentrations, and it occupies a greater volume. Moreover, the patterns of the distributions of concentrations of droplets are identical, indicating the insignificant influence of droplets on the gas flow velocity fields.

Figure 4 presents the calculated distributions of dimensionless volumetric concentrations of a liquid fuel c_f for cold spraying in the characteristic sections of a mixer: in the longitudinal section that passes through the center of the injection hole ($z = 0$) and in the transverse section immediately behind the injection hole at the distance $x = 0.28$ from

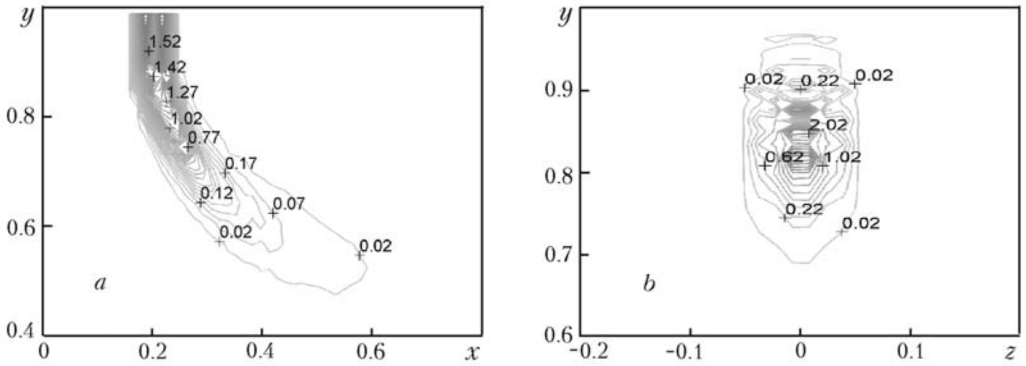


Fig. 3. Isolines of volumetric concentrations of fuel droplets in the central longitudinal (a) and transverse (b) sections of the mixer $x = 0.28$ with jetty supply of fuel (regime 1, $U_1 = 0$); $\alpha = 1.35$.

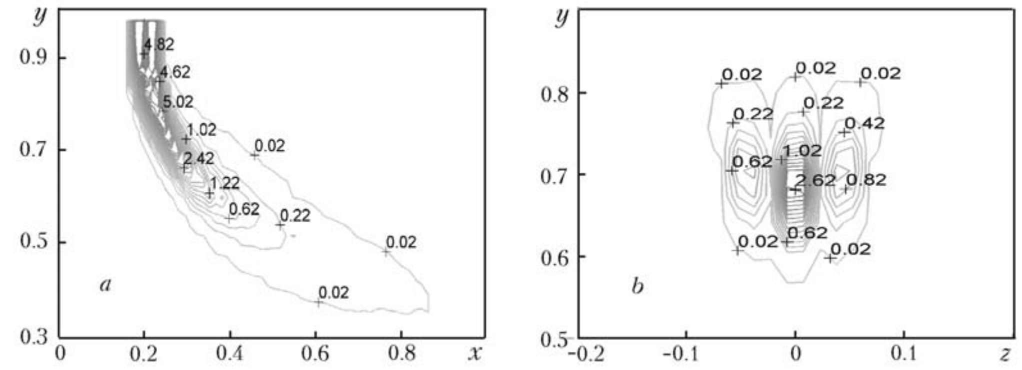


Fig. 4. Isolines of volumetric concentrations of fuel droplets in the central longitudinal (a) and transverse (b) sections of the mixer $x = 0.28$ with pneumatic supply of fuel; spraying by a cold air jet (regime 2, $U_1 = 20$ m/sec, $T_1 = 300$ K); $\alpha = 1.35$.

the inlet section. A comparison with Fig. 3 shows that in the case of pneumatic spraying the patterns of fuel distribution change appreciably. However, in this case too the influence of exchange by momenta between the air and droplets on the distribution of concentrations is hardly noticeable.

From the graphs of the distributions of the volumetric concentrations of fuel droplets it is seen that on the whole the latter follow the air flow. The splitting of the fuel jet in the transverse direction in pneumatic spraying is associated with the appearance of intense circulation flows in the wake of the spraying air jet. The absence of such splitting in jetty spraying indicates that the secondary flows induced by droplets are insufficiently intense. We note that when a high-temperature air jet is injected into a stalling flow, the depth of fuel penetration into a mixer is smaller than in the case of spraying by a cold jet (see Fig. 5). This effect is due, first of all, to the lessening of the penetrating power of an air jet (injection of a gas of a smaller density) and, second, to the enhancement of the processes of heating and evaporation of droplets in a high-temperature air flow of the injected jet.

Thus, in both jetty and pneumatic spraying of a fuel for the regimes considered it is possible to neglect the exchange of momenta between the gas and droplets and judge the interaction of droplets with the air flow from temperature fields. Quantitatively the intensity of heat transfer is characterized, firstly, by the dimensions of the region in which the gas temperature is smaller than that of the surrounding flow (in this case the boundary of this region is $T = 900$ K) and, secondly, by the minimum gas temperature in the computational domain. The former quantity indicates the part of the space where the air temperature underwent a change and the latter — the quantity of heat taken by droplets from the gas. The values of the minimum gas temperatures are given in Table 2 for all the operating conditions considered.

TABLE 2. Minimum Gas Temperature, K

Regime	Variant 1	Variant 2	Variant 3
1	638	539	447
2	300	300	300
3	724	612	502

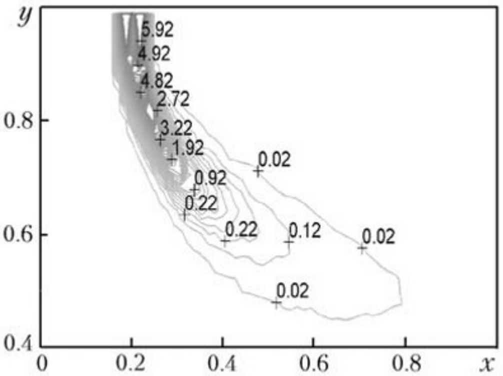


Fig. 5. Isolines of volumetric concentrations of fuel droplets in the central longitudinal (a) and transverse (b) sections of the mixer $x = 0.28$ with pneumatic supply of fuel; spraying by a hot air jet (regime 3, $U_1 = 20$ m/sec, $T_1 = 900$ K); $\alpha = 1.35$.

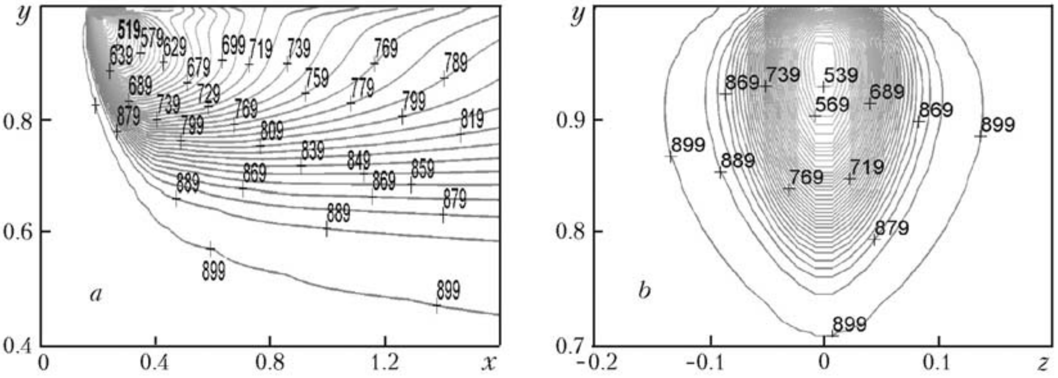


Fig. 6. Isolines of air temperatures in the central longitudinal (a) and transverse (b) sections of the mixer $x = 0.28$ with jetty supply of fuel (regime 1, $U_1 = 0$); $\alpha = 1.35$.

The calculations have shown that even in the absence of supply of the spraying air the gas temperature depends substantially on the values of operating conditions. The distributions of air temperatures in the absence and in the presence of a spraying air are presented in Figs. 6–8.

Figure 6 characterizes the direct influence of heat exchange between the gas and droplets on temperature fields, since in the absence of this exchange air has the same initial temperature over the entire region of flow. From the distributions of temperatures in the longitudinal sections of the model it is seen that at $\alpha = 1.35$ the region of heat transfer at $x = 1.6$ extends in the direction of the y axis to the distance $\Delta y = 0.55$. As calculations showed, at $\alpha = 5.4$ this distance is equal to $\Delta y = 0.42$. The minimum temperatures that correspond to these variants are equal to 447 and 683 K (Table 2). For the variant $\alpha = 2.7$ this quantity is equal to 539 K. Thus, on increase in the fuel flow rate through a jet injector the influence of droplets on temperature fields becomes more and more appreciable. As calculations show, on injection of a cold spraying air (Fig. 7), when heat transfer is mainly determined by the interaction of the main and spraying flows, this effect is virtually unnoticeable.

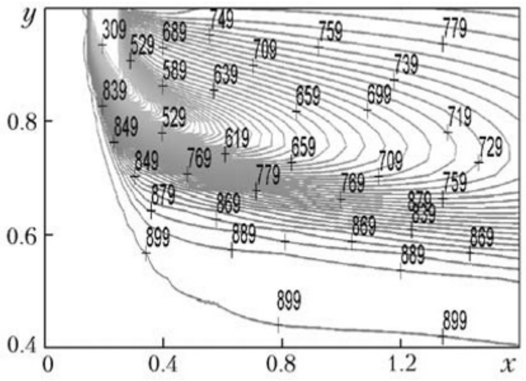


Fig. 7. Isolines of air temperatures in the central longitudinal section of the mixer; spraying by a cold air jet (regime 2, U_1 = 20 m/sec, T_1 = 300 K); α = 5.4.

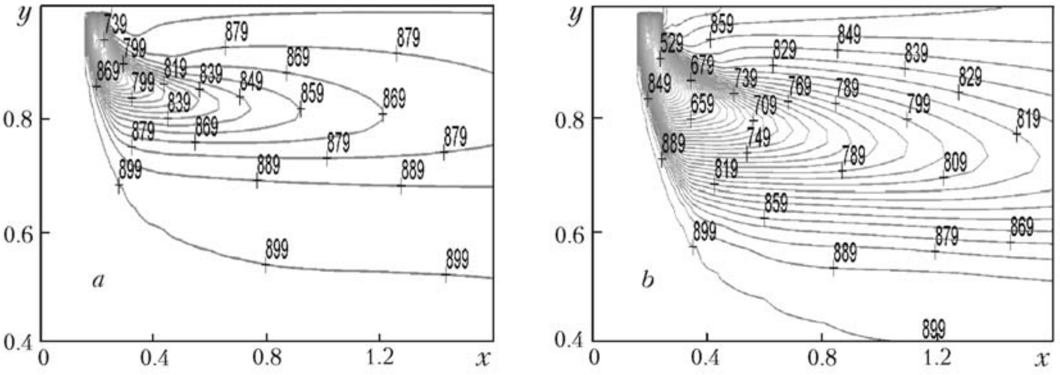


Fig. 8. Isolines of air temperatures in the central longitudinal section of the mixer; spraying by a hot air jet (regime 3, U_1 = 20 m/sec, T_1 = 900 K); a) α = 5.4; b) 1.35.

When a hot spraying air is injected (T_1 = 900 K), heat transfer will again be determined by the interaction of air flows with droplets, and therefore the influence of the fuel flow rate on the formation of temperature fields becomes appreciable (Table 2). The corresponding graphs are presented in Fig. 8. It is seen that in these cases the influence of droplets manifests itself virtually in the entire flow region.

Considering the model of heat transfer suggested in the present work, two moments must be noted. The first is that the change in the gas temperature occurs owing to the transfer of heat from the gas to droplets and is spent to heat and evaporate them. As calculations show, both latter processes are essential despite the fact that the basic fraction of droplets ($D_d < 100\ \mu$m) evaporates rather rapidly in the high-temperature air flow (T_1 = 900 K). The second moment is that heating and evaporation are the mechanisms that underlie heat transfer in the very gas phase and they are also two in number. The first is the conventional diffusion transfer of heat and the second — its convective transfer due to secondary flows which are either initiated by droplets or result from the flow of the stalling stream around the spraying air jets.

In the case of jetty supply of fuel the incipient secondary flows are of low intensity, and droplets are weakly entrained by such flows. This is expressed as the absence of individual vortex structures in the distributions of both concentrations and temperatures in the transverse sections of the module. The lowering of the gas temperature occurs exclusively at the expense of interphase exchange. Vortex structures are clearly seen in transverse sections with pneumatic spraying on the graphs of the distribution of fuel concentrations. A comparison between the distributions of temperatures and concentrations in these cases shows that the concentration profiles are much narrower than the corresponding temperature profiles in both longitudinal and transverse directions. This is associated with the intense diffusion heat fluxes, with the droplets mainly following the air flow. Attention is also drawn to the fact that the penetrating ability of a "cold" fuel-air jet is higher than that of a "hot" one due to the following two reasons: the great energy of the "cold" jet and the more intense process of heating and evaporation of droplets in the "hot" jet.

A comparison of gas cooling in spraying of a fuel by a hot air jet and in jetty spraying shows that although the fuel is injected into flows with identical temperatures, in the second case the lowering of the gas temperature is more appreciable. This seems to be due to the fact that on injection of droplets into a stalling air flow the velocity of droplets relative to the gas is higher than in the case of injection into a cocurrent flow. The rate of the evaporation of droplets is also higher and, consequently, the complete evaporation of droplets occurs over smaller distances and in smaller volumes, thus leading to the effect noted. The total quantity of heat transferred from air to droplets is the same in both cases, but the differences observed allow one to make different fuel-air mixtures by supplying a fuel either into a cocurrent air flow or into a stalling one.

Conclusions. Just as in jetty spraying, so in a pneumatic one, for the regimes considered it is possible to neglect the exchange of momenta between the gas and droplets and to judge the interaction of droplets with an air flow from temperature fields.

Injection of a fuel by a jet injector may cause a substantial change in the gas temperature. In the given case it occurs due to heat transfer from the gas to droplets and is spent on their heating and evaporation. In the case of pneumatic spraying of a fuel by a cold air jet the influence of interphase exchange is insignificant. Heat transfer is predominantly determined by the interaction of the main and spraying flows. During injection of a hot spraying air, when heat transfer inside the gas flow is less intense, the influence of the injection of a fuel on the formation of temperature fields again becomes appreciable. However, in this case the gas is cooled less than in jetty spraying. This effect is due to the fact that when droplets are injected into a stalling air flow, the rate of their evaporation is higher than during injection into a stalling flow.

The conclusions drawn confirm the necessity of taking into account the processes of interphase heat and mass exchange in numerical models and emphasize that the characteristics of fuel-air mixtures may substantially differ depending on whether a fuel is supplied into a cocurrent air flow or into a stalling one.

This work was supported by the Russian Fundation for Basic Research, projects No. 05-0817942 and No. 07-08-00573.

NOTATION

C_f, volumetric concentration of a liquid fuel, kg/m^3; c_f, coefficient of specific heat of liquid, J/(kg·K); $c_{p\mathrm{g}}$, coefficient of specific heat of gas at constant pressure, J/(kg·K); C_R, coefficient of droplet resistance; C_v, concentration of fuel vapor per unit volume, kg/m^3; D_d, droplet diameter, m; H, channel height, m; h, specific total enthalpy, J/kg; k, energy of turbulence per unit mass, m^2/sec^2; L, latent heat of evaporation, J/kg; m_d, mass of a droplet, kg; m_f, mass fraction of kerosene vapors; $\mathrm{Pr} = \mu_\mathrm{g} c_{p\mathrm{g}}/\lambda_\mathrm{g}$, Prandtl number; $\mathbf{R}$, force of aerodynamic resistance; $\mathrm{Re} = \rho_\mathrm{g} D_\mathrm{d} W/\mu_\mathrm{g}$, Reynolds number of a droplet; S_φ, internal source term in the equation of transfer of the variable φ; T, temperature, K; t, time, sec; $\mathbf{U}_\mathrm{g}$, vector of averaged gas velocity; $U_{\mathrm{g}i}$ ($i = 1, 2, 3$), components of the vector of averaged gas velocity, m/sec; $\mathbf{V}_\mathrm{d}$, vector of droplet velocity; $\mathbf{W} = \mathbf{V}_\mathrm{d} - \mathbf{U}_\mathrm{g}$, vector of droplet velocity relative to gas; x, y, z, Cartesian coordinates; α, summed coefficient of air excess; Γ_φ, coefficient of diffusion transfer of variable φ; Δt_d, time of droplet residence in the volume element, sec; Δv, elementary volume, m^3; ε, rate of dissipation of turbulence energy, m^2/sec^3; λ_g, thermal conductivity of gas, W/(m·K); μ_g, coefficient of dynamic viscosity of gas, kg/(m·sec); ρ, density, kg/m^3; φ, dependent variable. Subscripts and superscripts: 0, main flow; 1, spraying air; g, gas; f, liquid fuel; d, droplet; int, interphase; v, vapor-like fuel; i, individual droplet.

REFERENCES

1. Yu. V. Polezhaev (Ed.), *Combustion Laws* [in Russian], Énergomash, Moscow (2006).
2. V. A. Skibin and V. I. Solonin (Eds.), *Contribution to the Creation of Aviation Engines* [in Russian], Books 1 and 2, Mashinostroenie, Moscow (2000).
3. B. G. Mingazov, *Combustion Chambers of Gas-Turbine Engines. Construction, Modeling of Processes, and Calculation* [in Russian], Kazansk. Gos. Tekh. Univ., Kazan' (2004).

4. A. I. Maiorova, A. A. Sviridenkov, and V. V. Tret’yakov, Influence of interphase exchange on mixture formation in the module combustion chamber, *Vestn. Samarsk. Gos. Aérokosm. Univ. im. akad. S. P. Koroleva*, No. 2, 71–75 (2002).
5. A. B. Lebedev and V. V. Tret’yakov, *Methods of Calculation of Fuel Distribution in Combustion Chambers*, Review No. 355, Izd. TsIAM, Moscow (1992).
6. S. Patankar, *Numerical Methods for Solving Problems of Heat Transfer and Fluid Dynamics* [Russian translation], Énergoatomizdat, Moscow (1984).
7. K.-J. Chien, Predictions of channel and boundary-layer flows with low-Reynolds-number turbulence model, *AIAA J.*, **20**, 33–38 (1982).
8. V. V. Tret’yakov, Calculations of the concentrations of a sprayed fuel in combustion chambers, in: V. I. Yagodkin (Ed.), *Separation Flows in Combustion Chambers*, *Trudy TsIAM*, No. 1203, 96–106 (1987).
9. Yu. F. Dityakin, L. A. Klyachko, B. V. Novikov, and V. I. Yagodkin, *Spraying of Liquids* [in Russian], Mashinostroenie, Moscow (1977).